HEAT TRANSFER COEFFICIENTS IN VARIOUS STRENGTH GRAVITATIONAL FIELDS

A THESIS

Presented to

the Faculty of the Division of Graduate Studies

Georgia Institute of Technology

In Partial Fulfillment

of the Requirements for the Degree

Master of Science in Mechanical Engineering

By

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December 1956

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Bryan Bell Marsh, Jr.

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Approved:

Date Approved by Chairman: 5 November 1956

ACKNOWLEDGMENTS

For the conception of this thesis and the valuable guidance and suggestions extended during its completion, I am deeply indebted to Dr. Thomas W. Jackson. I wish also to express appreciation to the other members of my committee, Dr. Charles W. Gorton and Dr. Henderson Ward, for their helpful suggestions and criticism of the report. I would like to thank Miss Shelby Smith for her help in editing and typing the first draft of the report and Mrs. Wanda Reaves for preparing the final manuscript and tables. The efforts of all these people represent a significant contribution to the completion of the project, and their efforts are all sincerely and gratefully acknowledged.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

SUMMARY

The purpose of this investigation was to gather and examine heat transfer data for superimposed forced and free convection in various strength gravitational fields.

A search of existing literature reveals that very little work has been done with heat transfer in various strength gravitational fields. A correlation for superimposed forced and free convection in laminar flow through a horizontal pipe has been established that is applicable to a rotating apparatus when the forced convection velocities are at right angles to the free convection velocities.

An appropriate rotating apparatus was designed and constructed to examine the effects of varying gravitational forces on water as the heat transfer medium. After the instruments were calibrated, test runs in both the laminar and the transitional range were conducted.

An analysis of the data revealed that considerable error was introduced, in all probability, by the errors inherent in the rotating thermocouple pick up. The data in the laminar range were inconclusive as far as added heat transfer effects due to induced free convection were concerned. However, the data did indicate a correlation with the expression given for superimposed forced and free convection in a horizontal pipe.

In the transitional flow range the data indicated that increased gravitational strengths yielded higher Nusselt numbers. As the Reynolds numbers increased, the system moved from a mixed flow region to a forced flow region where the forced convection was predominant.

It was recommended that the apparatus be redesigned and the work extended.

CHAPTER I

INTRODUCTION

Interest in the study of heat transfer in various strength gravitational fields increased with the development of gas turbine power. The cooling problems encountered in gas turbines as well as those inherent in aircraft and rocket development added to the pursuit of information concerning the effects of gravitational force on the mechanisms of heat transfer. It is extremely important in the solutions of these problems that the effects of gravitational force be evaluated from actual data.

Great progress in the study of heat transfer in various strength gravitational fields was made when it was realized that the gravitational field generated in the rotating components of propulsion systems often created strong free convection currents. Also, it was found that turbine blades could be effectively cooled by free convection currents alone. (1)* In fact Schmidt (2) actually proposed a cooling system utilizing free convection in rotating turbine blades containing a liquid coolant. His conclusion was that the heat from combustion gases flowed into the cooling fluid, which filled the blade passages, creating large temperature differences and thus causing rather intense free convection currents. The Grashof number, which represents the free convection, has high values in this case due to the temperature difference and centrifugal force.

Eckert and Diagulia (3) found in their literature search that very little experimental data existed on free convection in the Grashof number

*Numbers in parentheses refer to references listed in bibliography.

range above 10^{12}. They conducted experiments with air flowing in a vertical tube of length-to-diameter ratio of 5.0 with a Grashof number range from 10^{8} to 10^{13}. Their data generally correlated satisfactorily with the expression of Nusselt number as a function of Grashof and Prandtl numbers. For laminar flow the expression used to correlate their data was

$$Nu = 0.416 \left[Gr \quad Pr \right]^{1/4} \tag{1}$$

and for turbulent flow it was

$$Nu = 0.0252 \left[Gr \quad Pr \right]^{2/5} \tag{2}$$

Martinelli and Boelter (4) obtained an approximate solution of the heat transfer and hydrodynamic equations for combined forced and free convection with viscous flow in a vertical tube at uniform well temperature. In this case the free convection currents, which are a function of the gravitational field, are in the same direction as the forced flow. Their analysis yield the equation as

$$Nu_{am} = 1.75\, F_1 \sqrt[3]{Gz_m + 0.0722\, F_2 \left(\frac{Gr\ Pr\ D}{L} \right)_w^{0.84}} \tag{3}$$

They also state that the problem of forced and free convection in a horizontal tube would be much more complex since the free convection velocities would act at right angles to the forced convection velocities. For heating in a horizontal tube the fluid seems to pass down the tube in a helical path, rising at the sides and falling at the center. The result of this observation is an equation for laminar forced convection on which is superimposed a free convection flow due to the gravitational field. This formula which utilizes the magnitude of the vector sum of the forced and free convection velocities is given as

$$Na_{am} = 1.75 \;\; F_1 \;\; \sqrt[6]{\left[Gz_m\right]^2 + \left[0.0722 \;\; F_2 \; \left(\frac{Gr \;\; Pr \;\; D}{L}\right)_w^{0.84}\right]^2} \qquad (4)$$

Martinelli, Southwell, Alves, Craig, Weinberg, Lansing, and Boelter (5) analyze several sets of data and find that the equation of Martinelli and Boelter for forced and free convection in a vertical pipe predicts the experimental data within twenty per cent. They also mention, as do Martinelli and Boelter in reference four, that the equation for flow in a horizontal tube seems to have merit but needs further study.

In a report concerned with forced and free convection in turbulent flow through a vertical pipe, Eckert, Diaguila, and Livingood (6) analyzed the data obtained by Eckert, Diaguila, and Curren (7) as well as that of Clark and Rohsenow (8). They found that, for large values of Grashof and Prandtl numbers, the Nusselt numbers converge into a free convection region. For decreasing values of Grashof and Prandtl numbers, the Nusselt number becomes independent of the Grashof and Prandtl numbers and the flow enters a forced-flow region. They indicate that the flow may be divided into three regions; the forced flow region in which the free convection is negligible, the free flow region where the free convection practically alone determines the heat transfer, and the mixed flow region in which both free and forced convection are of equal importance.

An analytical investigation incorporating the forced and free convection effects in the cooling of gas turbine blades was made by Jackson and Livingood (9). They concluded that, in addition to giving increased heat transfer effects, the high free convection forces encountered in the rotating turbine wheel and blades would also generate a tremendous pumping force.

It is interesting to note that in most of the previous experimental work performed, the tests were run in vertical tubes. High Grashof numbers were obtained by using large dimensions and temperature differences. Also, very little experimental work has been done on forced and free convection in a horizontal pipe.

It is the object of this paper to present data obtained in a rotating apparatus from which the Nusselt numbers can be calculated as functions of Grashof and Prandtl numbers. The large centrifugal forces obtained in rotation will allow an examination of the effect of gravitational forces on heat transfer. Since the data will be reduced in terms of Grashof numbers as well as "g" forces, the results should be applicable to horizontal pipe conditions. This may lead to a correlation of the obtained data with the expression given by Martinelli and Boelter (4) for laminar flow in a horizontal pipe.

CHAPTER II

EXPERIMENTAL PROCEDURE

In order to obtain heat transfer data in varying strength gravitational fields, it was necessary to construct a rotating heating section. Heat was transferred to water which flowed through the apparatus. The inlet and outlet water temperatures as well as the mass rate of water flow were measured. With this information, it was possible to calculate the rate of heat flow into the water.

A temperature sensing device was devised to determine the wall temperature of the rotating heated tube in which the water was flowing. This wall temperature along with the mean fluid temperature, the physical dimensions of the heated section, and the heat flow gave sufficient information to calculate the heat transfer coefficient. The heat transfer coefficient was reduced to the Nusselt number and its dependence upon Reynolds, Prandtl, and Graetz numbers was shown.

To assure laminar flow, the flow rate had a maximum value of approximately 20 pounds per hour. The temperature rise of the water was kept above a minimum value of 10 degrees Fahrenheit to minimize the possible error.

The gravitational strength was varied over a range of values to a maximum value of approximately 600 g's. The Grashof number was calculated knowing the gravitational strength, physical dimensions, fluid properties, and the required temperature difference. This information was assembled in graphical form with Nusselt number shown as a function of Graetz, Reynolds, Grashof, and Prandtl numbers in an appropriate form.

CHAPTER III

DESCRIPTION OF APPARATUS

General.--An apparatus to study the effects of centrifugal force on heat transfer necessitates a rotating heating section with supporting equipment to monitor temperatures, flow rates, and power inputs. The rotating component was supported on a 13 inch shaft made from a stainless steel tube with an outside diameter of 9/16 inches and an inside diameter of 5/16 inches. A piece of 1/4 inch outside diameter and 0.089 inch inside diameter stainless steel tubing was circled around the shaft with a 7.5 inch coil diameter. The two ends were brought down and welded in holes two inches apart on the same side of the shaft. A drum of mild steel with a removable face plate nine inches in diameter was welded onto the shaft in such a way that one leg of the tube coil was located outside the drum while the loop and other end were inside. (See Figures 1 and 2.) The leg outside, or the cold leg, was located 4-1/4 inches from the end of the shaft while the leg inside the drum, or the hot leg, was 6-3/4 inches from the other end of the shaft. Two brass plugs were silver soldered in the shaft between the two legs with an insulating air gap separating them. This allowed the fluid to enter the shaft at the cold end, proceed out through the cold leg into the loop, return to the shaft through the hot leg, and flow out the hot end of the shaft.

Heat source.--Heat was applied to the water in the loop portion of the tubing by means of a 640 watt resistance wire heater. The heater was coiled around the tube forming the loop and was insulated from it by a layer of asbestos paper and glass cloth. (See Figure 3.) The whole drum was packed with insulation to minimize heat losses through the drum to the outside air.

The power supply to the heater was transmitted to the rotating apparatus by a system of copper brushes and rings. A lucite shield served both as an insulator and a mounting base for the rings while the brushes were mounted on a rod pivoted in a tube atop of the shaft bearing housing. (See Figures 4 and 5.) The whole assembly was kept in constant contact by means of a spring attached to the rod holding the brushes.

Bearings and seals.--The shaft was machined for two inches on each end to accommodate the 9-3L01 New Departure Ball Bearings. Tailstocks of two wood lathes were modified to serve as bearing housings and were mounted, facing each other, on one of the lathe beds. The inner parts of the tailstocks were easily adapted to function as packing glands thus preventing leakage of water from the system. (See Figure 5.)

Speed variation.--The drive for the rotating apparatus consisted of a three-phase electric motor, a variable speed pulley, and a two inch pulley mounted on the shaft, all of which were connected by one-half inch Vee belts. The motor was mounted on a separate lathe bed and the two beds were connected by a frame made from two inch angle iron. The variable speed pulley was mounted on the angle iron frame in line with the motor and shaft pulley. (See Figure 6.) The variable speed pulley was essentially two pulleys mounted on the same shaft with a floating face separating them. Adjustments were made with a hand wheel which moved the pulley in line with the shaft and motor. The resulting force on the belts caused the floating face to move back and forth changing the radii of the pulleys, hence changing the speed of the shaft.

Thermocouples.--Three number 30 gage copper-constantan thermocouples were used to measure the wall temperature of the coil. Two of these were spot-welded on the wall on the heated portion of the loop section while the other was spotted on the hot leg of the tube coming into the shaft. The thermocouple wires were protected by small stainless steel tubes leading them in towards the shaft from the loop. The signals were transmitted from the rotating shaft by means of rings and brushes similar to the power leads. (See Figures 4 and 5.) The thermocouples had a common constantan lead thus requiring only four pickups. The brushes were mounted on a pivoted rod, similar to the power transmitting assembly, which was linked to the front panel board. They were so arranged that, during normal operations, the brushes were not in contact with the rings. The brushes were closed only while taking readings.

Thermocouples of 30 gage copper-constantan wire were inserted into each end of the shaft by means of probes to measure the inlet and outlet water temperatures. They were located approximately at the points where the water entered the cold leg of the small tube and where the water returned to the shaft out of the hot leg. The junctions were embedded in 1/8 inch diameter plugs of silver solder. For details of these probes see Figure 5.

Instrumentation.--The electrical regulating and measuring system consisted of a Powerstat Variable Transformer, type S649, manufactured by Superior

Electric Company, which adjusted the power input to the heater. This regulated power was measured by a single phase wattmeter, model 310, manufactured by Weston Electric Company.

Rotational speed of the apparatus was measured with a Strobatac, type number 631-B, made by the General Radio Company.

The thermocouple leads from the apparatus were brought into a Leeds and Northrup Selector Switch. From the switch, two leads proceeded to an ice point and then to a Portable Precision Potentiometer, number 8662, also made by Leeds and Northrup.

Flow rates were measured by two different means depending on whether the flow occurred in the laminar or turbulent range. For laminar flow the water was discharged into an open container and weighed. In the turbulent range a Stabl-Vis Flowrater, manufactured by Fischer and Porter, was used.

For location of instruments used in the apparatus see Figure 7 in the Appendix.

CHAPTER IV

PERFORMANCE OF TESTS

Thermocouple calibration.--After the apparatus had been completed and test runs had been made, it was evident that some means of calibration would be necessary to compensate for the induced emf in the rotating thermocouple pickups. The losses were due to the combination of convective cooling and frictional heating at the junctions formed by the rings, brushes, and leads.

The calibration was obtained by allowing the apparatus to rotate open to ambient air. It was noticed that, at any given speed, the error reached an equilibrium point and became steady after a period of twenty minutes. Thus, the error could be found as a function of speed and ambient temperature provided the apparatus was allowed to run for twenty minutes before readings. The calibration curves are given as Figure 8 in the appendix.

Test procedure.--In the actual test runs, the first critical variable, namely the water flow rate, was controlled through a valve between the building distribution system and the apparatus. When the flow rate was in the laminar range, the water flowed from the apparatus through a needle valve into an open container where it was weighed for a set period of time. For the turbulent range, the water flowed through the Flowrater into the drain.

After the flow rate had been set, the power was gradually applied to the heater by varying the voltage through the Powerstat transformer. The actual power level of the system would depend on the amount of heat required to assure at least a ten degree temperature rise in the water.

When a sufficient temperature rise was reached, the motor driving the apparatus was started. The apparatus was allowed to rotate at the first speed of 760 revolutions per minute for a period of at least twenty minutes to allow for the thermocouple calibration and to assure steady state in the system.

When the system reached equilibrium, readings were made of the inlet and outlet water temperatures, the wall temperatures, the ambient air temperature, the voltage and power to the heater, the flow rate, the pressure in the system, rotational speed, and the barometric pressure. In reading the wall temperatures which were transmitted from the rotating apparatus by means of the ring and brush assembly, it was first necessary to close the contacts to complete the circuit. The contacts were left closed only for the time required for the reading, in order that the induced error would be kept at a minimum.

With the first run completed, the rotational speed was increased by means of the variable speed pulley to successive values of 1200, 1800, and 2400 revolutions per minute. At each of these settings, the same procedure was followed. After data had been taken for the previously set flow rate at the four different speeds, the flow rate was changed to a new value and tests were run at the same speeds in the same manner. From preliminary calculations, it was found that for laminar flow, the rate should be lower than 20 pounds per hour and for turbulent flow, it should range between 50 and 120 pounds per hour. The upper limit for turbulent flow was due to the restriction of a ten degree rise in water temperature.

CHAPTER V

ANALYSIS OF EXPERIMENTAL RESULTS

Assumptions.--Upon completion of the test runs in the laminar flow range, it was evident that some assumptions must be made concerning the tube wall temperatures. The reason for this decision was that the tube wall temperatures appeared lower than the exit water temperature. At first it was thought that the probe measuring the exit water temperature might be rubbing inside of the rotating shaft and the resultant frictional heating would explain the high temperatures. A series of tests run at different speeds with water standing in the apparatus and with no heat applied indicated only a slight temperature rise as measured by the probes. This slight rise was attributed to frictional heating of the water and the theory of the probes rubbing in the shaft was discarded. With this possibility eliminated, the source of error was attributed to the rotating thermocouple pickups.

In understanding the assumptions made, it is necessary to examine (see Figure 3) the locations of the thermocouples on the tube wall. The thermocouples, numbers two and three, read the temperature of the heated section of the tube while thermocouple number four read the wall temperature of the hot leg returning to the shaft. The first assumption made was that at point four, the water and tube wall are in a state of thermal equilibrium with the temperature of each being the same as the exit water temperatures. This seems to be a valid assumption since the exit water probe is located at the point where the water re-enters the shaft. Also, a calculation indicated that the temperature gradient between the inside of the tube wall and the outside wall was less than a degree and could be neglected.

The second assumption came as a result of realizing that the error present in the rotating thermocouple pickup would be identical for all of the three tube wall temperatures. An average value of the heated wall temperatures--that is, thermocouples numbers two and three, was taken. Then the difference in the average heated wall temperature and the temperature of the hot leg, which was assumed in equilibrium with the water, was computed. The second assumption made was that this difference when added to the exit water temperature would give the average value of the heated wall.

An examination of the data in the transitional range revealed that the wall temperature was running higher than the exit water temperature as it normally should. Justification of the two assumptions in this range was realized when the effect of heat conduction through the shaft to the rings was considered. In the laminar range, the exit water temperature was considerably higher than in the transitional range, thus there would be a greater temperature gradient through the shaft wall. The result was that more heat was conducted to the thermocouple pickup. The effect of this additional heat in the thermocouple circuit was that a negative emf was introduced and gave lower temperature readings than actually existed. With the temperature gradient reduced in transitional flow, the negative emf was also reduced but not completely eliminated. Therefore, although the wall temperatures seemed higher than the exit water temperatures, the readings were still lower than the actual values.

Calculations.--Values of the physical properties of water used in this analysis were taken from Eckert (11). A tabulation of values used at the various temperatures is given in the appendix. In the calculations of actual quantities found in the experimental systems, properties were evaluated at the arithmetic mean temperature of the water. In equations, where properties were evaluated at the wall temperatures, the subscript w will appear.

The experimental results are given in the appendix in tabular form as well as being represented in figures. A detailed discussion on the equations used in calculations is also given in the appendix.

Discussion of results.--A study of Figures 9 through 11 indicates that there is considerable scatter in the points. In the laminar range, the points are in general agreement with the correlation as proposed by Martinelli and Boelter (4) but there is a lack of a definite trend. No concrete conclusion can be drawn from the data as to whether increasing the Grashof number by rotation has any effect on the Nusselt number.

In the transitional range where the flow is approaching turbulent flow, there is an effect of increasing Grashof number noticed by most of the points. (See Figures 10 and 11.) The points indicate that at higher Grashof numbers, the Nusselt numbers are slightly higher. Also, it is noticeable that the trend of the points indicates that the flow is approaching a forced flow region, as proposed by Eckert (6), where the effects of forced convection are predominant. This theory is substantiated both by the fact that the points at higher Grashof numbers are approaching the points at low and zero Grashof numbers, and by the fact that all of the points after seemingly converging, approach a correlation for forced flow. The forced flow equation by McAdams (10) is higher than the experimental data for a zero Gr Pr D/L ratio. This, as well as the trend in the laminar region, indicates a systematic error in the system which tends to make the experimental data lower than would be expected. The equation of forced turbulent flow is not generally applicable in the transitional range, but it was used here as a reference since the Reynolds numbers indicated that the flow was approaching turbulence.

It is likely that had higher Grashof numbers been obtained in the experiments, the effect of the free convection would be much more noticeable. It is very probable that all of the points lie either in the forced flow region or just barely in the mixed flow region.

Sources of error.--The validity of some of the experimental points is questionable and can be accounted to several sources of error. At some points it is doubtful if steady state had actually been reached before readings were made. As has already been previously mentioned, the rotating thermocouple pickups and the water temperature probes could also be a source of error affecting the results. The most probable source of error encountered was discovered when on two occasions, a small leak developed in the weld holding the small tube in the shaft. The water from this leak saturated the insulation in the drum and, since the saturation was not uniform, this no doubt had some effect on the tube wall temperature. Twice

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

The following conclusions appear logical and generally justifiable on the basis of the preceding presentation:

(1) The equation of Martinelli and Boelter for laminar, superimposed forced and free convection in horizontal pipes is appropriate for correlation of data in a rotating apparatus where the forced convection velocity is perpendicular to the free convection velocity.

(2) The theory proposed by Eckert of a forced flow region, mixed flow region, and a free convection region depending on the predominant mode of convection is essentially fact and merits further investigation.

(3) An increase in gravitational strength due to rotation does have the effect of increasing the heat transfer coefficient when the flow is in either the mixed flow region or the predominant free convection region.

It is highly recommended that this investigation be extended to include Grashof number ranges of 10^3 to 10^{15}. In this wide range, the experimental data should include all three of the regions as proposed by Eckert. The investigation should also be extended to Reynolds numbers of the magnitude of 10^6 with other fluids, possibly liquid metals as well as water.

In continuing the experiments for water, it is recommended that the apparatus be redesigned. Changes to be considered are a larger diameter coil with provisions for studying length-to-diameter effects. A more reliable driving mechanism, and a more substantial frame is needed to reduce vibration. A more accurate and reliable temperature sensing device is needed both for the wall temperatures as well as the water temperature measurements. A suitable heater to allow a larger temperature rise in the water should be used.

It is hoped that work following this will profit by the mistakes and limitations set forth in this report.

APPENDIX A

NOMENCLATURE

NOMENCLATURE

Capital Letters

A	Ares, ft^2
D	Diameter, ft
F_1	Function of Nu/Gz due to use of arithmetic mean temperature difference
F_2	Function of Nu/Gz due to buoyant force acting on fluid becoming smaller as fluid approaches tube-wall temperature
L	Length, ft
R	Radial length from centerline of shaft to centerline of tube loop

Lower Case Letters

c_p	Specific heat, Btu/lb-°F
g	Acceleration of gravity, ft/sec^2
h	Heat transfer coefficient, Btu/hr ft^2 °F
k	Thermal conductivity, Btu/hr-ft-°F
l	Length, ft
q	Heat transfer rate, Btu/hr
r	Radius, ft
t	Temperature, °F
w	Weight flow of fluid, lb/hr

NOMENCLATURE (Cont'd)

Greek Letters

β	Coefficient of expansion of fluid, $°F^{-1}$
μ	Dynamic viscosity, lb-sec/ft^2
ν	Kinematic viscosity, ft^2/sec
ρ	Density, lb/ft^3

Subscripts

m	Condition at mean fluid property
w	Condition at tube wall
h	Passage for heat transfer
b	Passage for pressure drop
2, 3, 4, 6, 7	Thermocouple locations

Dimensionless Moduli

Gz	Graetz, $4wc_p/\pi kL$
Gr	Grashof, $gD^3\beta(t_w - t_m)/\nu^2$
Nu	Nusselt, hD/k
Pr	Prandtl, $c_p\mu/k$
Re	Reynolds, $4w/\mu\pi D$

APPENDIX B

FIGURES

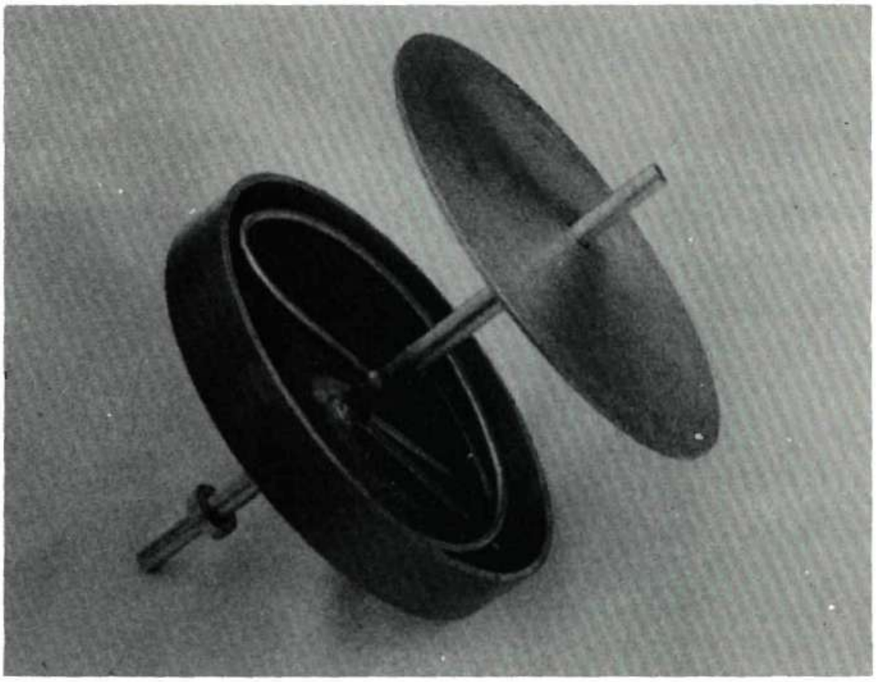

Figure 1. Photograph Showing Position of Tube Loop in Drum.

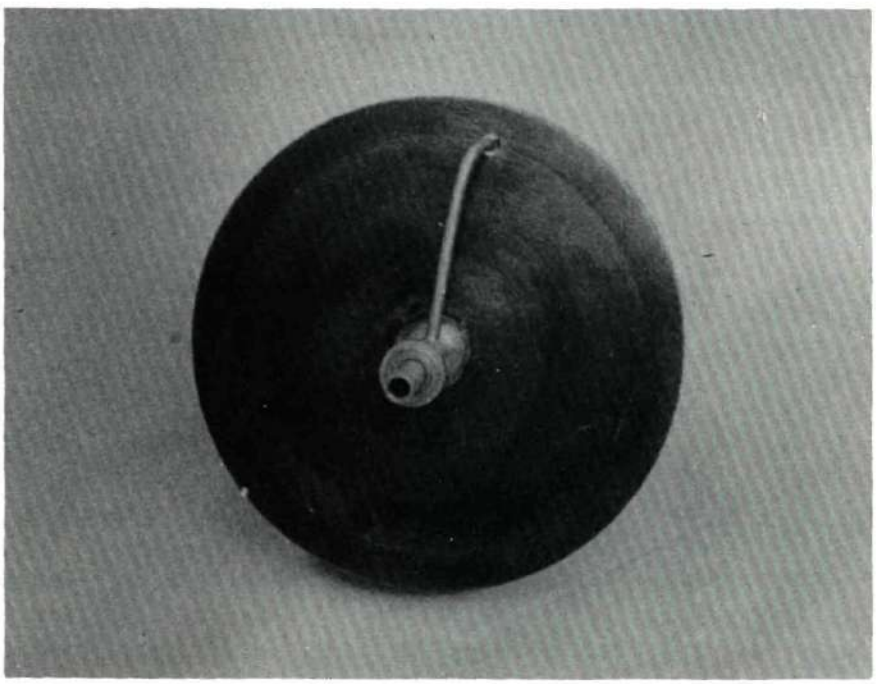

Figure 2. Photograph of Drum and Cold Leg Outside.

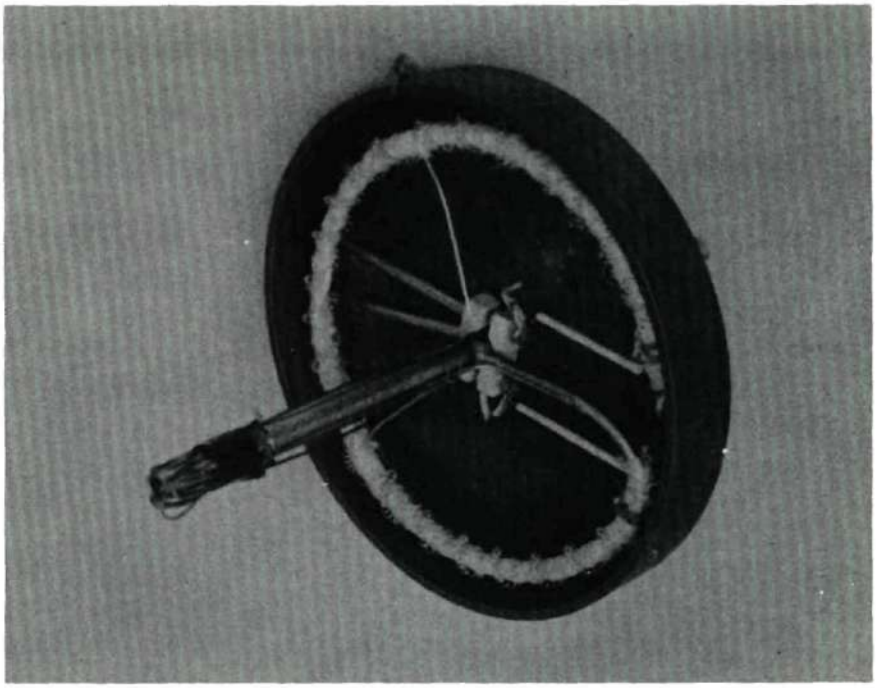

Figure 3. Photograph of Coiled Heater and Wall Thermocouples.

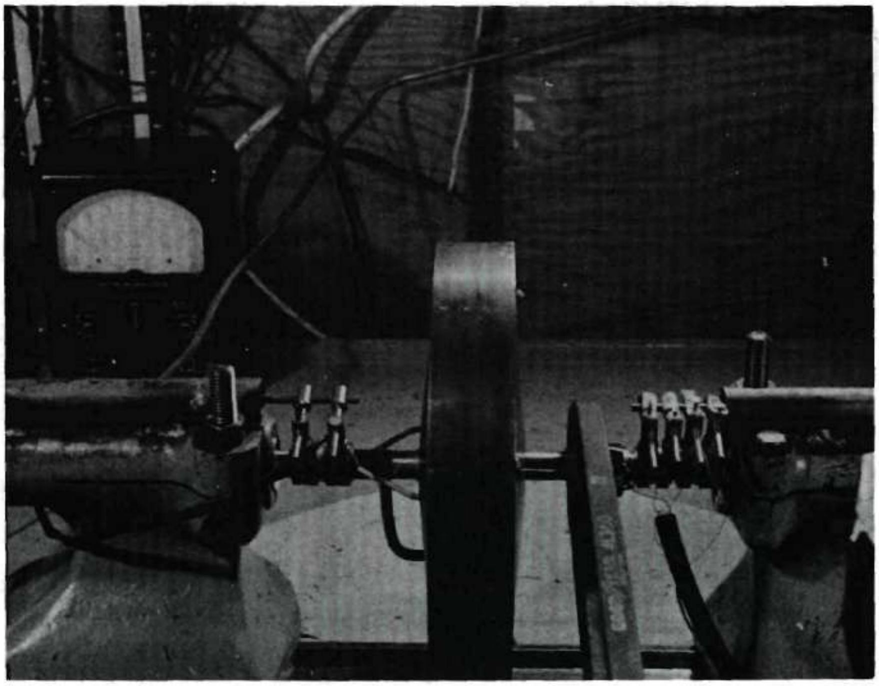

Figure 4. View of Assembled Thermocouple Pickups and Apparatus.

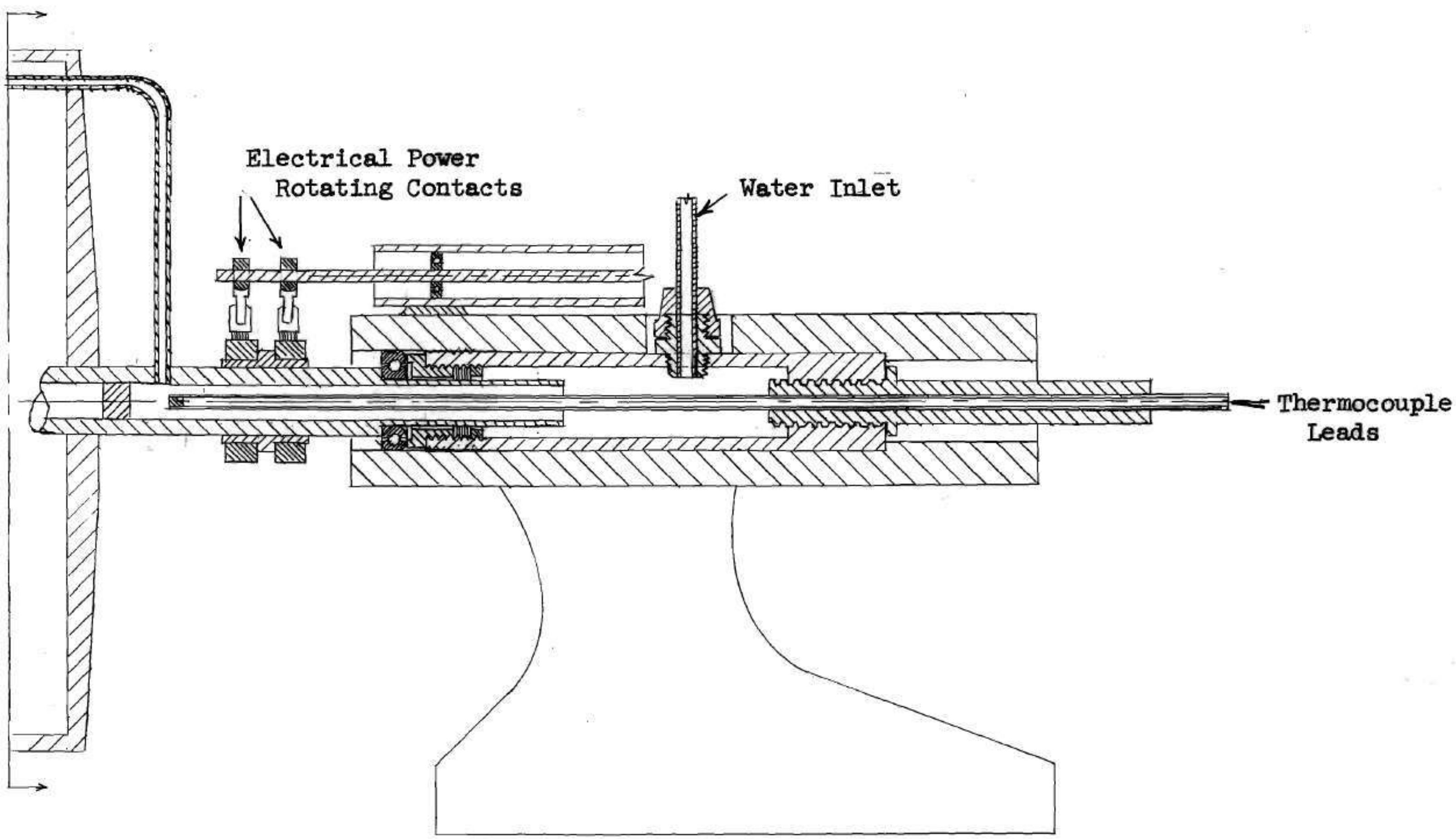

Figure 5. Detail View of Bearing Housing, Water Probe and Slip Ring Arrangement

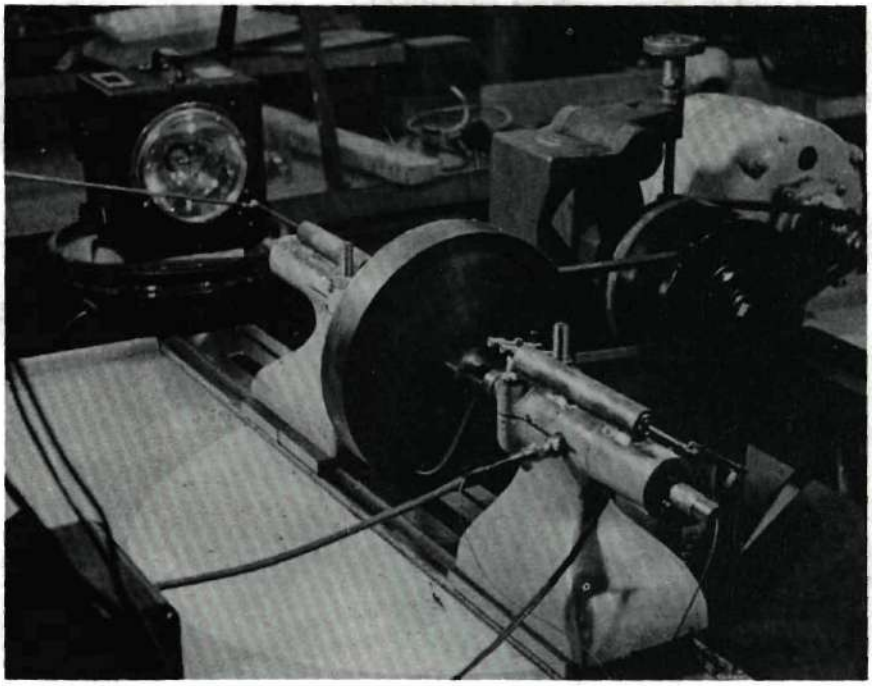

Figure 6. View of Assembled Apparatus.

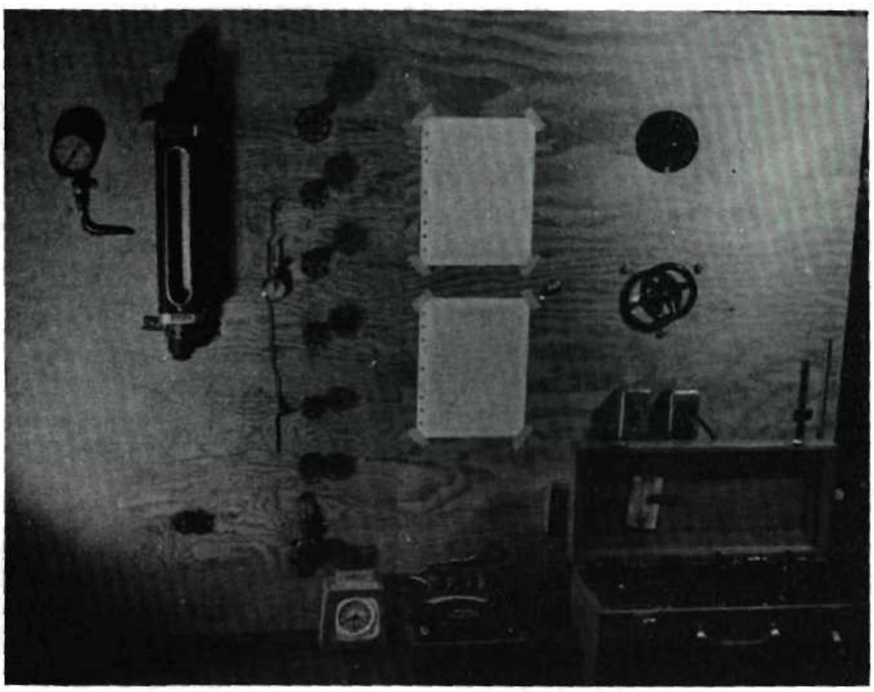

Figure 7. View of Front Panel Board.

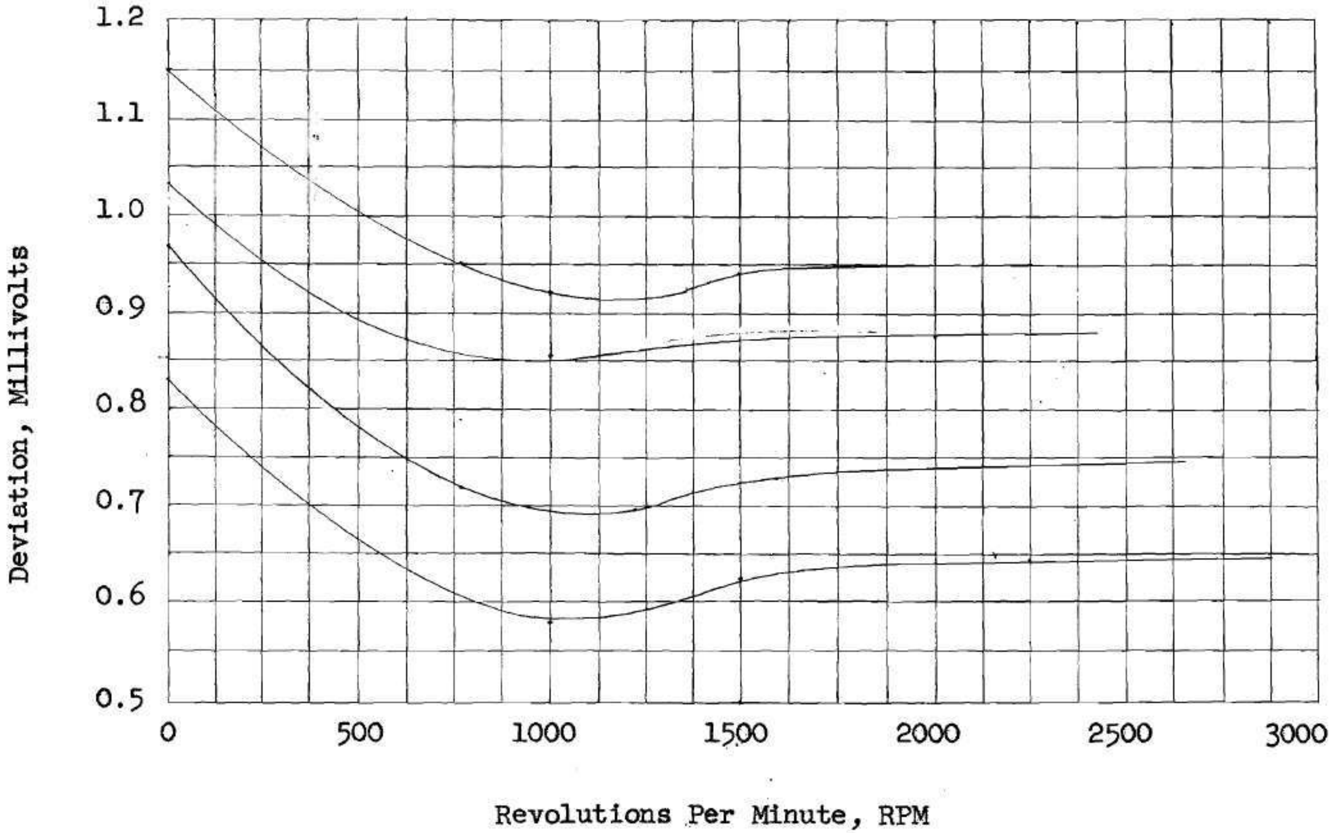

Figure 8. Rotating Thermocouple Calibration Curves

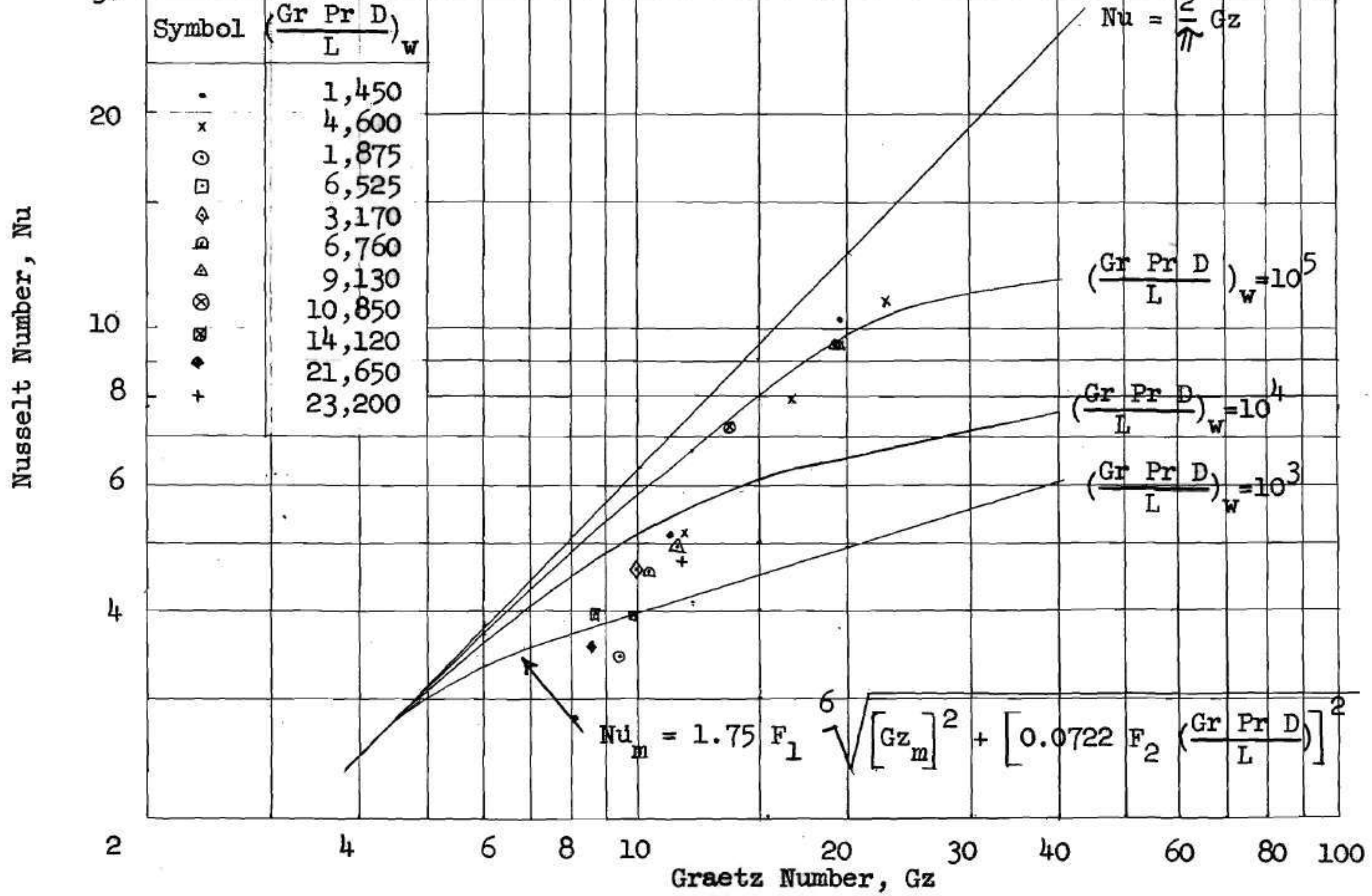

Figure 9. Experimental Values of Nusselt Number in Laminar Forced and Free Convection

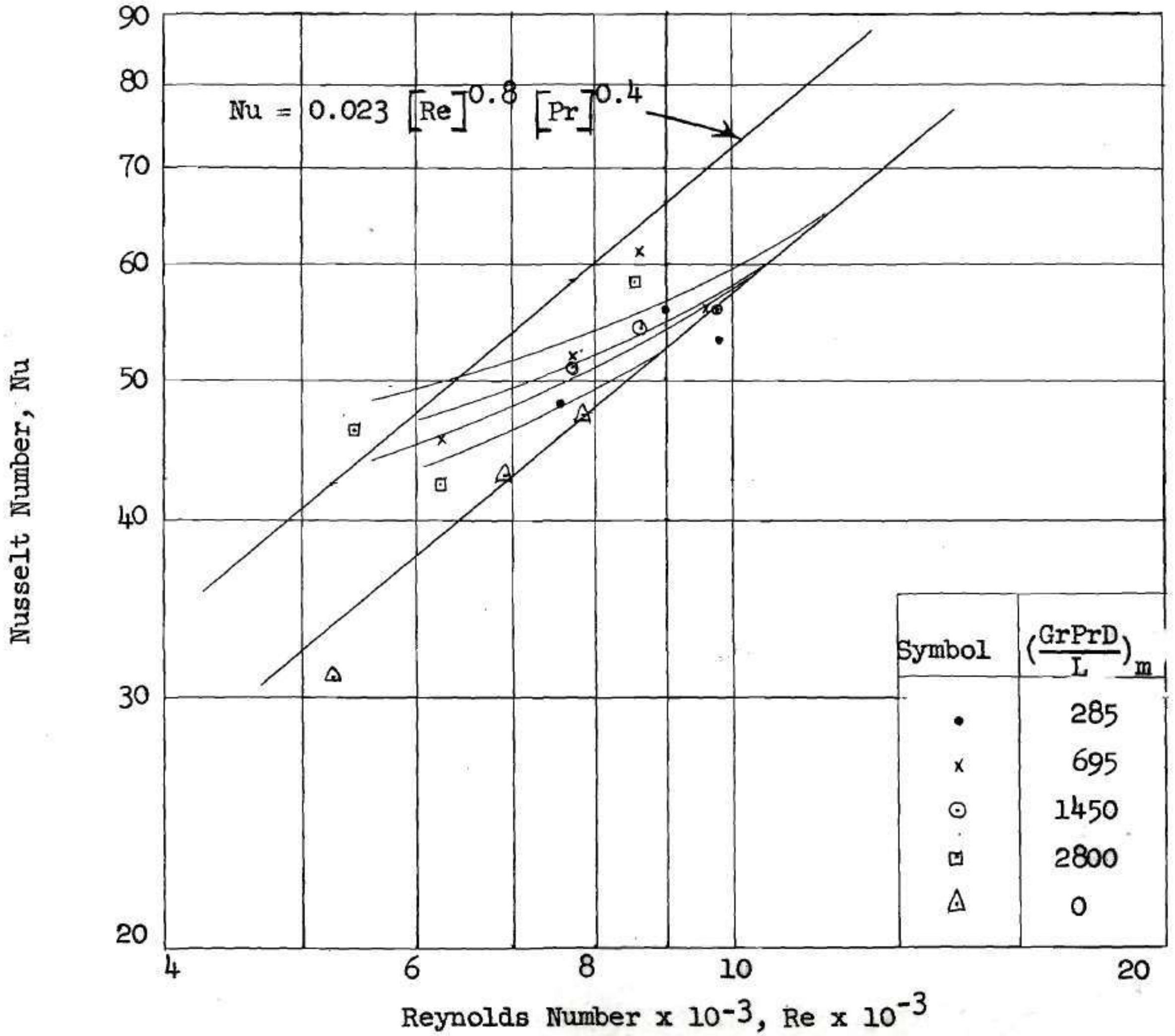

Figure 10. Experimental Values of Nusselt Number in Transitional Forced and Free Convection

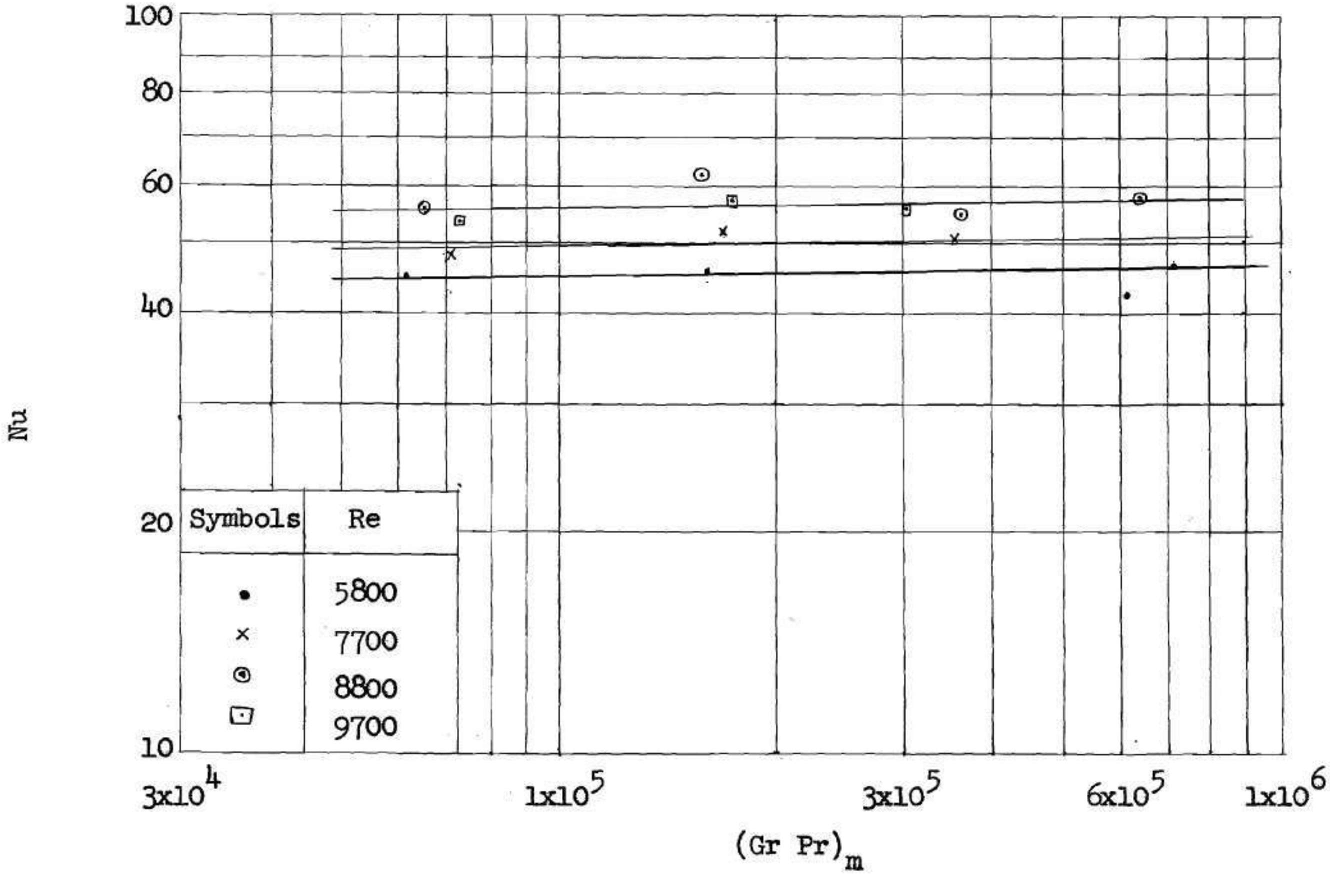

Figure 11. Experimental Values of Nusselt Number in Transitional Forced and Free Convection With Reynolds Number Parameter

APPENDIX C

TABLES

Table 1. Calculated Results

Run No.	t_w	$t_w - t_m$	Nu_m	Re_m	Gz_m	$Gr_m \times 10^{-4}$
1	148.0	44.3	3.42	760	9.46	9.22
2	126.9	36.6	2.84	543	8.04	5.59
3	133.8	41.6	5.39	765	11.10	6.56
4	128.2	35.7	10.35	1318	19.93	5.18
5	143.7	42.9	7.95	1234	16.68	19.64
6	145.5	49.3	5.40	837	11.43	18.76
7	158.0	59.5	3.96	679	8.63	29.0
8	125.3	30.6	4.57	674	9.78	13.89
9	127.6	32.1	4.54	721	10.24	29.80
10	130.8	38.7	3.95	713	9.91	60.30
12	141.6	41.0	4.96	839	11.08	42.20
13	131.8	29.1	7.26	978	13.64	49.40
14	120.5	24.7	10.90	1512	22.55	20.75
15	124.6	26.1	9.43	1345	19.50	41.20
16	124.7	25.5	9.51	1318	19.07	40.60
17	154.7	50.5	4.69	933	11.52	108.70
18	186.2	70.8	3.58	827	8.42	122.50
19	93.2	10.9	45.0	5830	107.0	1.11
20	92.9	11.4	45.6	6280	116.4	2.86
21	93.7	12.9	42.5	6220	116.9	10.77
22	95.7	12.3	46.3	5440	101.4	12.91
23	90.9	10.7	58.5	8670	164.2	11.14
24	91.4	11.5	54.4	8690	165.5	6.23
25	92.1	11.8	51.8	7740	139.9	2.95
26	93.1	12.6	48.3	7560	142.5	1.23
27	92.8	13.3	53.0	9730	186.5	1.26
28	93.4	13.6	55.7	9680	185.0	3.22
29	89.3	8.1	73.8	9160	171.0	0.
30	109.1	20.9	31.0	5240	90.9	0.
31	94.5	11.8	56.0	9000	164.4	1.18
32	93.3	10.9	61.0	8790	161.2	2.85
33	93.9	12.3	56.0	9780	181.2	5.37
34	97.8	13.9	51.1	7750	139.0	6.51
35	100.7	16.5	43.0	6890	123.1	0.
36	97.5	15.0	47.5	7840	143.7	0.

Table 2. Physical Properties and Calculated Results

Run No.	$(\frac{Gr\ Pr\ D}{L})_m$	$\nu_w x10^6$	$Gr_w x10^{-4}$	Pr_w	$(\frac{Gr\ Pr\ D}{L})_w$	$(Gr\ Pr)_m$ $x\ 10^{-4}$
1	1,477	4.79	15.73	2.81	1,875	
2	1,061	5.71	9.20	3.42	1,334	
3	1,223	5.39	11.74	3.19	1,587	
4	1,009	5.63	9.23	3.36	1,315	
5	3,415	4.96	36.10	2.92	4,460	
6	3,290	4.89	36.25	2.87	4,405	
7	4,730	4.43	60.0	2.57	6,525	
8	2,590	5.79	21.6	3.47	3,170	
9	5,430	5.69	47.1	3.39	6,760	
10	10,790	5.53	101.5	3.28	14,120	
12	7,190	5.03	72.6	2.97	9,130	
13	8,760	5.48	78.8	3.25	10,850	
14	3,970	6.03	31.2	3.63	4,790	
15	7,710	5.83	61.2	3.49	9,050	
16	7,570	5.82	61.8	3.49	9,140	
17	17,230	4.50	207	2.65	23,200	
18	16,020	3.71	245	2.09	21,650	
19	262					6.19
20	682					16.10
21	2,600					61.30
22	3,000					70.95
23	2,710					64.0
24	1,525					36.1
25	716					16.95
26	298					7.04
27	310					7.34
28	790					18.66
29	0					0
30	0					0
31	277					6.55
32	672					15.86
33	1,285					30.2
34	1,500					35.5
35	0					0
36	0					0

Table 3. Calculated Results

Run No.	Flow Rate lb/hr	Temp. Rise of Water	q Btu/hr	Per Cent q to Water	RPM	No. of g's
1	6.16	49.7	306		760	61.4
2	5.18	39.9	206.5	53.5	762	61.7
3	7.17	60.9	436	66.9	762	61.7
4	12.87	57.0	733	73.5	762	61.7
5	10.80	63.3	683	68.5	1215	156.1
6	7.40	72.1	533	66.0	1119	132.8
7	5.60	85.0	475	71.1	1183	148.8
8	6.32	44.2	279	62.0	1192	150.9
9	6.60	44.3	291	63.7	1810	848
10	6.40	47.0	306	67.5	2295	557
12	7.16	57.4	409	63.5	1780	336
13	8.82	48.0	422	64.2	2400	612
14	14.58	36.7	533	70.8	1810	347
15	12.60	39.0	490	63.5	2410	616
16	12.32	39.2	483	62.9	2415	618
17	7.50	66.5	497	62.8	2430	627
18	5.59	93.5	521	65.3	1840	359
19	64.8	14.7	953		760	
20	73.8	13.7	1009		1206	
21	73.8	14.4	1061		1750	
22	62.6	17.7	1108		2410	
23	103.6	11.7	1211		2487	
24	104.3	11.6	1210		1806	
25	92.4	12.8	1182		1220	
26	90.0	13.1	1178		760	
27	117.5	11.6	1361		760	
28	116.2	12.6	1462		1197	
29	108.0	10.8	1165		0	
30	57.0	22.3	1270		0	
31	104.3	12.3	1282		750	
32	102.5	12.6	1288		1215	
33	115.2	11.6	1336		1590	
34	88.8	15.6	1382		1600	
35	78.6	17.6	1381		0	
36	91.2	15.2	1384		0	

Table 4. Actual Data

Run No.	Ambient Temperature	Indicated Wall Temperatures t_2	t_3	t_4	Inlet Water Temp.	Outlet Water Temp.
1	80.9	121.6	123.6	120.8	86.5	146.2
2	66.9	106.8	106.8	106.3	74.3	126.4
3	66.9	113.8	113.8	112.7	71.7	132.6
4	67.1	117.3	117.3	115.3	69.7	126.7
5	68.7	127.8	127.9	124.5	77.0	140.3
6	68.5	122.7	122.7	120.8	71.6	143.7
7	69.2	125.5	125.5	124.7	72.2	157.2
8	77.3	110.1	111.3	109.7	79.7	124.5
9	77.5	109.7	111.2	109.0	81.9	126.2
10	77.9	112.0	113.4	111.5	82.6	129.6
12	78.2	120.0	122.0	118.9	82.2	139.6
13	78.0	115.6	117.5	114.5	81.8	129.8
14	79.5	114.5	116.5	112.5	81.0	117.7
15	79.6	116.3	118.4	114.3	82.6	121.6
16	79.7	117.1	119.4	115.6	82.8	122.0
17	79.3	127.3	129.6	124.4	83.9	150.4
18	79.2	142.0	145.7	140.7	90.1	183.6
19	72.3	102.2	102.2	101.0	74.9	89.6
20	71.8	107.1	108.3	105.8	74.6	88.3
21	71.7	110.5	110.5	106.0	74.6	89.0
22	71.5	110.2	111.0	107.1	74.5	92.2
23	70.6	110.9	112.7	107.0	74.3	86.0
24	70.4	112.4	114.2	107.4	74.1	85.7
25	69.8	114.3	116.1	109.9	73.9	86.7
26	69.5	112.6	114.0	107.0	73.9	87.0
27	69.2	114.1	116.3	108.1	73.7	85.3
28	68.8	119.1	120.5	113.3	73.5	86.1
29	77.2	96.6	96.6	95.5	75.8	86.6
30	77.4	113.8	112.5	103.2	77.0	99.3
31	78.2	124.3	126.5	119.9	76.5	88.8
32	78.9	127.3	128.2	123.1	76.1	88.7
33	78.7	123.1	127.1	118.5	75.8	87.4
34	78.2	124.8	127.7	120.1	76.1	91.7
35	74.7	115.3	117.4	108.7	75.4	93.0
36	74.2	114.2	116.3	107.8	74.9	90.1

Table 5. Physical Properties

Run No.	Mean Water Temp.	$\nu_m \times 10^6$	$(c_p)_m$	k_m	ρ_m	Pr_m
1	116.4	6.25	0.9986	0.3677	61.9	3.78
2	100.4	7.32	0.9985	0.3615	62.1	4.49
3	102.2	7.20		0.3622	62.1	4.40
4	98.2	7.50		0.3604	62.1	4.60
5	108.7	6.73		0.3644	62.0	4.11
6	107.7	6.80		0.3640	62.0	4.15
7	114.7	6.36		0.3672	61.9	3.86
8	102.1	7.20		0.3622	62.1	4.41
9	104.1	7.04		0.3630	62.1	4.31
10	106.1	6.90		0.3639	62.1	4.22
12	110.9	6.58		0.3658	62.0	4.01
13	105.8	6.93		0.3637	62.1	4.24
14	99.4	7.41		0.3610	62.2	4.54
15	102.1	7.20		0.3622	62.1	4.41
16	102.4	7.18		0.3623	62.1	4.40
17	117.2	6.21	0.9986	0.3682	61.9	3.75
18	136.9	5.25		0.3750	61.5	3.09
19	82.3	8.93		0.3530	62.3	5.57
20	81.5	9.00		0.3525	62.3	5.63
21	80.8	9.10		0.3522	62.3	5.70
22	83.4	8.81		0.3536	62.3	5.49
23	80.2	9.15		0.3519	62.3	5.75
24	79.9	9.20		0.3517	62.3	5.79
25	80.3	9.14		0.3519	62.3	5.74
26	80.5	9.12		0.3520	62.3	5.72
27	79.5	9.24		0.3515	62.4	5.82
28	79.8	9.21		0.3516	62.3	5.80
29	81.2	9.03		0.3524		5.66
30	88.2	8.33		0.3558		5.27
31	82.7	8.88		0.3531		5.55
32	82.4	8.92		0.3530		5.57
33	81.6	9.00		0.3525		5.63
34	83.9	8.77		0.3537		5.45
35	84.2	8.73		0.3539		5.42
36	82.5	8.90		0.3530		5.56

APPENDIX D

CALCULATIONS

EQUATIONS USED IN CALCULATIONS

Heat input to water

$$q = w\ c_p\ (t_7 - t_6)$$

Actual nusselt number

$$q = h\ A\ (t_w - t_m)$$

$$Nu = \frac{hD}{k} = \frac{D}{k}\left[\frac{q}{A\ (t_w - t_m)}\right]$$

$$Nu = \frac{q}{k\pi L\ (t_w - t_m)} \qquad \text{where } L = 21 \text{ in.}$$

EVALUATION OF DIMENSIONS AND CONSTANTS YIELDS

$$Nu = 0.182\ \frac{q}{k\ (t_w - t_m)}$$

Reynolds number

$$Re = \frac{4w}{\mu\pi D} = \frac{4w}{\nu\rho\pi D} \qquad \text{where } D = 0.089 \text{ in.}$$

EVALUATION OF DIMENSIONS AND CONSTANTS YIELDS

$$Re = 0.0477\ \frac{w}{\nu\rho}$$

Graetz number

$$Gz = \frac{Wc_p}{kL} = \frac{\pi}{4}\ Re\ Pr\ \frac{D}{L}$$

$$Gz = 0.00329\ Re\ Pr$$

Grashof number

$$Gr = \frac{g\ D^3 \beta\ (t_w - t_m)}{\nu^2}$$

But

$$g = R\omega^2 = R\left[\frac{2\pi \times RPM}{60}\right]^2$$

EVALUATION OF g AND D^3 YIELDS

$$Gr = 14.1 \times 10^{-10}\ \frac{\beta\ (t_w - t_m)(RPM)^2}{\nu^2}$$

Average Wall Temperature

$$\frac{t_2 + t_3}{2} - t_4 = \Delta t_w$$

This calculation performed in millivolts. Δt_w converted to degrees in region of t_7.

$$t_w = t_7 + \Delta t_w$$

BIBLIOGRAPHY

BIBLIOGRAPHY

1. Diagulia, Anthony J., and Freche, John C., "Blade-to-Coolant Heat-Transfer Rates and Operating Data from a Natural-Convection Water-Cooled Single-Stage Turbine." N. A. C. A. Research Memorandum E51117, 1951.

2. Schmidt, E., "The Possibilities of the Gas Turbine for Aircraft Engines." Reports and Transcript No. 489, GDC 2504 T, British M. C. S.

3. Eckert, E. R. G., and Diaguila, A. J., "Experimental Investigation of a Free-Convection Heat Transfer in Vertical Tube at Large Grashof Numbers." N. A. C. A. Report 1211, 1955.

4. Martinelli, R. C., and Boelter, L. M. K., "The Analytical Prediction of Superposed Force and Forced Viscous Convection in a Vertical Pipe." Press Publications in Engineering, University of California, Berkeley, California, Vol. 5, No. 2, pp. 28-85, 17 figures in text.

5. Martinelli, R. C., Southwell, C. J., Avles, G., Craig, H. L., Weinberg, E. B., Lansing, N. F., and Boelter, L. M. K., "Heat Transfer and Pressure Drop for a Fluid Flowing in the Viscous Region through a Vertical Pipe." Transcript of the American Institute of Chemical Engineers, Vol. 38, No. 3, June 25, 1942.

6. Eckert, E. R. G., Diaguila, Anthony, J., and Livingood, John, N. B., "Free-Convection Effects on Heat Transfer for Turbulent Flow Through a Vertical Tube." N. A. C. A., TN 3584, December, 1955.

7. Eckert, E. R. G., Diaguila, Anthony J., and Curren, Arthur N., "Experiments on Mixed-Free-and-Forced-Convective Heat Transfer Connected with Turbulent Flow Through a Short Tube." N. A. C. A., TN 2974, 1953.

8. Clark, John A., and Rohsenow, Warren M., "Local Boiling Heat Transfer to Water at Low Reynolds Numbers and High Pressures." Div. Ind. Cooperation, N. I. T., July, 1952. See also: Transcript, American Society of Mechanical Engineers, Vol. 76, No. 4, May, 1954, pp.553-562.

9. Jackson, Thomas W., and Livingood, John N. B., "Analytical Investigation of Two Liquid Cooling Systems for Turbine Blades." N. A. C. A., Research Memorandum, E51F04, August, 1951.

10. McAdams, William H., Heat Transmission, 3rd ed., New York: McGraw-Hill Book Co., Inc., 1954.

11. Eckert, E. R. G., Introduction to the Transfer of Heat and Mass, 1st ed., New York: McGraw-Hill Book Co., Inc., 1950.